18 pts.

SUPERCHARGING SYSTEM FOR GAS TURBINES

Applicant claims the benefit of co-pending U.S. provisional patent application Serial No. 60/138,848 filed June 10, 1999; Serial No. 60/139,894 filed June 22, 1999; Serial No. 60/152,277 filed September 3, 1999; Serial No. 60/159,207 filed October 13, 1999; and Serial No. 60/195,302 filed April 10, 2000. This application is a continuation-in-part of co-pending patent application Serial No. 09/388,927, filed September 2, 1999.

BACKGROUND OF THE INVENTION

Field of the Invention

This invention relates generally to gas turbine power generation systems and more particularly relates to a supercharging system for improving the capacity of gas turbine power plants at high ambient temperatures. Specifically, the system uses a supercharging fan combined with a controller to pressurize inlet air to the turbine to allow operation with existing turbines; the supercharging fan is preferably combined with an inlet air cooling system.

Background and Prior Art

It has long been recognized that the capacity of gas turbines declines with increasing inlet air temperature; the typical penalty is on the order of 0.4% per degree F, and this relationship is illustrated in Figure 1. This characteristic is especially troublesome for gas turbines used in electrical power generation since the peak electricity demand usually coincides with the highest ambient air temperatures. Gas turbines and

associated generators and power distribution systems are usually rated based on turbine capacity at 40 to 50 °F inlet air temperature. This low rating temperature means that the capacity reduction at summer-peaking conditions can amount to approximately 20 to 40% of turbine capacity, depending on the design, local weather conditions, and the characteristics of the particular turbine.

Many different approaches for cooling inlet air to the turbine in order to reduce or eliminate this capacity penalty are known in the prior art. A summary of these approaches is described in the ASME paper, "Options in Gas Turbine Power Augmentation Using Inlet Air Chilling," Igor Ondryas et al., presented at the Gas Turbine and Aeroengine Congress and Exposition, June 11-14, 1990, Brussels, Belgium. Among the alternatives for cooling are direct and indirect evaporative cooling, electric vapor-compression, absorption, and thermal storage systems.

Of these many alternatives, direct evaporative cooling is the only approach that has seen any significant commercial application. Direct evaporative cooling has the advantage of low cost and simplicity, but the ambient wet-bulb temperature limits the possible temperature reduction. For locations in the eastern U.S., direct evaporative cooling can reduce inlet air temperatures by 10 to 20 °F, depending on the local climate. Larger reductions are possible in warm, dry climates such as those of the southwestern U.S. While direct evaporative cooling is helpful, it does not allow the turbines to run at their full design capacity. After over 50 years of intensive research and development in gas turbines, a better approach for dealing with high ambient temperatures has not been produced.

An interesting but virtually unused approach to address these problems is described in the paper "Supercharging of Gas Turbines by Forced Draft Fans With Evaporative Intercooling" by R.W. Foster-Pegg, ASME 1965. This paper describes the use of a high-pressure fan to increase the inlet pressure to a gas turbine combined with an evaporative cooler downstream of the fan as a way of increasing turbine capacity. This approach could give large theoretical advantages, but it required special sizing of the generator which limited its use to new turbines. In addition, the systems used a single fan with inlet vanes for control purposes, which reduced the efficiency of the system.

Kolp et al. show the economics of the supercharging and cooling systems in the ASME paper "Advantages of Air Conditioning and Supercharging an LM6000 Gas Turbine Inlet," Journal of Engineering for Gas Turbines and Power, July 1995, vol. 117, p. 513-527. This paper shows that while evaporative cooling is extremely attractive, the economics of supercharging are not very attractive, with payback periods of over 10 years for simple cycles. Supercharging is more attractive in combined-cycle plants, but its economics are still marginal. The supercharging systems described in this paper are virtually identical to those from the 1960s, so supercharging has not advanced significantly despite decades of turbine development.

One significant problem in the prior art is that the supercharging arrangements require increasing the size of the associated generator and other auxiliary equipment. The cost of replacing the generator and other auxiliaries is so large that it effectively eliminates this possibility for existing plants. Even in new installations, supercharging may not be a practical option except at the very beginning of the project since the basic

requirements of the generator, power distribution system, and associated hardware would have to change. As Kolp et al. state in their paper (page 520), "in contrast to supercharging, it is not necessary to increase the size of gas turbine plant equipment when adding evaporative cooling." Thus the conventional wisdom is that evaporative cooling may be added to an existing power plant, but supercharging is not a retrofit option.

Another method for controlling turbine capacity involves a variable-speed compressor. Relevant patents include U.S. Patent Nos. 3,853,432 and 2,693,080. These systems would provide a large range of control and usually were intended for use in aircraft applications. A major problem with these systems is the cost and complexity of the variable-speed compressor. Related problems are the reliability and maintenance related to the large gearing required for these systems. These systems have not seen significant use in power-generation applications.

The use of fogging for cooling inlet air for gas turbines is an additional related technology in the prior art. For example, see Meher-Homji and Mee, *Gas Turbine Power Augmentation by Fogging of Inlet Air*, Proceedings of the 28th Turbomachinery Symposium, 1999. In addition to the benefit associated with cooling the inlet air, fogging can further improve turbine performance by cooling air inside the compressor. This intercooling effect is roughly a 5% increase in capacity for a water mass flow equal to 1% of the air mass flow. This paper shows that fog intercooling may also improve compressor efficiency.

One limitation of fogging is the amount of water mist that compressor may safely ingest. Excessive water can cause problems with corrosion, erosion, or other

damage to the compressor section of the turbine. At least one turbine manufacturer has expressed concern about the effect on the compressor and will not guarantee its turbines with fogging systems in many cases. While fogging offers some additional advantages compared to evaporative pads, concerns about potential adverse affects on compressor performance limit the capacity benefits of fogging to roughly 10% or less for many turbines in humid coastal climates.

SUMMARY OF THE INVENTION

In accordance with one preferred embodiment of the present invention, a system for improving gas turbine capacity at high ambient air temperatures includes at least one supercharging fan that pressurizes turbine inlet air and a controller which limits maximum turbine capacity to that of the turbine without supercharging. The system preferably includes an air cooler for reducing turbine inlet air temperature. The system of the invention may be added to an existing gas turbine or designed for a new, supercharged, gas turbine system.

One of the principal objectives and advantages of the invention is the ability to increase turbine capacity under summer-peaking conditions while reducing the installed cost of the system based on capacity at summer-peaking conditions. Related objectives of the invention are to create a relatively compact system that is not excessively complex and that does not create unacceptable reliability or maintenance problems. An additional objective is to develop a supercharger that allows practical retrofit of existing turbines. In accordance with another aspect of the present invention, a supercharger for a gas turbine comprises a fan and a fogger located in the inlet air stream of the turbine.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a graph illustrating the relationship between turbine capacity and ambient temperature;

Figures 2A-2C are schematic diagrams of one preferred embodiment of the invention that includes an air cooler downstream of a supercharging fan which pressurizes air entering a gas turbine;

Figure 3 shows another preferred embodiment of the invention that uses an indirect evaporative cooler;

Figure 4 shows an alternate embodiment of the invention that uses an axial-flow fan;

Figure 5 shows another alternate embodiment of the invention that uses series fans;

Figure 6 is a graph comparing the temperature-variant turbine capacity of several different turbine systems;

Figures 7A and 7B are schematic compressor maps that show the principles of operation of the system according to the invention;

Figures 8A and 8B are fan curves that show how multiple fans can be used to vary inlet pressure to a gas turbine;

Figure 9 is a graph of maximum supercharging pressure as a function of turbine inlet air temperature;

Figure 10 shows an alternate embodiment of the invention in which heated compressor output air is fed back into the compressor inlet air stream to regulate gas

turbine performance;

Figure 11 shows an alternate embodiment of the invention in which turbine output is controlled by regulating burner output;

Figure 12 shows an alternate embodiment of the invention in which turbine output is controlled by regulating compressor inlet temperature;

Figure 13 illustrates an embodiment of the invention employed in a combined-cycle (gas turbine/steam turbine) power plant;

Figure 14 illustrates a preferred embodiment of the invention which uses a cooling tower and a cooling coil to cool air leaving the supercharging fan;

Figure 15 is a perspective view illustrating the preferred construction configuration of high-pressure ductwork for use with the present invention;

Figures 16A and 16B are an end view and a section view, respectively, illustrating construction of a diffuser/flow accelerator section of ductwork constructed as shown in Figure 15;

Figures 17A and 17B are perspective views illustrating application of the ductwork configuration shown in Figures 15, 16A, and 16B to a supercharging fan assembly; and

Figure 18 is a schematic diagram of another preferred embodiment of the invention, wherein a supercharger for a gas turbine contains a fan and a fogger located in the inlet air stream of the turbine.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

Figure 2A shows a preferred embodiment of the invention. A gas turbine power

plant 11 contains a gas turbine system 10 and a generator 28. Gas turbine system 10 includes a compressor 12, a burner 14, and a turbine 16. The turbine 16 shares a common shaft 24 with the compressor 12 and the generator 28. The compressor 12 receives compressor inlet air stream 18 and pumps it to a higher pressure to create pressurized burner inlet air stream 20 which is supplied to burner 14. Burner 14 heats the air from stream 20 and supplies the resulting heated turbine inlet air stream 22 to the turbine 16. Turbine 16 rotates in response to the received heated inlet air stream 22, thereby rotating shaft 24 which in turn rotates generator 28 to generate electric power. Exhaust air stream 26 exits the turbine 16. This exhaust stream 26 may go directly into the atmosphere, or it can enter a steam boiler in the case of a combined cycle plant as described below in connection with Figure 13.

While Figure 2A depicts a simple turbine arrangement for the purpose of explaining the invention, it will be recognized that more complicated arrangements (such as a dual-spool configuration) also may be employed and do not change the principle of operation of the invention. Of course, the gas turbine will normally include filters, controls, safety devices, etc. as is known to those skilled in the art, and as such no illustration or detailed explanation of such components is provided or necessary for purposes of explanation of the present invention.

The inlet air stream 18 is provided to the compressor 12 as follows. A first supercharging fan 30 and a second supercharging fan 32 draw ambient air 40 and supply pressurized air to plenum 38. The air then goes through air cooler 34 which cools the air, thereby forming a compressor inlet air stream 18 that enters compressor 12. The air

cooler 34 is preferably a direct evaporative cooler that cools and humidifies the air stream. Examples of possible configurations for evaporative coolers are well known in the prior art. Other alternatives for the air cooler include a direct-expansion evaporator, a chilled water coil or direct-contact heat exchanger, an indirect evaporative cooler, or other device for lowering the temperature of the air stream. For the case of a chilled-water coil, cold water can be provided by a vapor-compression chiller, an absorption chiller, or from naturally occurring sources of cold water such as groundwater or bottom water from deep lakes or seas. In the case of an absorption chiller, waste heat from the turbine exhaust may be used as a heat source to drive the chiller. The air cooler is preferably located in the air stream between the supercharging fans and the turbine, but it also could be located upstream of the supercharging fans. The advantage of locating the air cooler downstream of the supercharging fans is that it can remove any heat added by the supercharging fans.

A bypass damper 36 allows air to enter the plenum 38 without going through the supercharging fans when the supercharging fans are not operating. First supercharging fan 30 has a corresponding first damper 42 on its discharge end. Likewise, second supercharging fan 32 has a second damper 44 on its discharge end. The controller 50 receives an input control signal from sensor 52 and controls operation of supercharging fans 30 and 32 in accordance with such signal. Possible control inputs include air stream temperature, compressor outlet pressure, generator output power, and ambient air temperature. The controller can be as simple as a thermostat; alternatively, it may include computer (i.e., microprocessor) control and other associated electronics that also may control and monitor turbine performance.

Dampers 42 and 44 and bypass damper 36 act as check valves to prevent reverse air flow from the gas turbine. The dampers preferably open in response to a pressure gradient across the damper, with a gravity return to the closed position in the absence of a pressure gradient. Figures 2B and 2C show how the dampers operate in response to different fan operating modes, as explained more fully below.

The first and second supercharging fans 30 and 32 preferably are belt-driven centrifugal fans or direct-drive axial fans. For centrifugal fans, the preferred design uses backwardly inclined airfoil blades to maximize efficiency. Fans of this type can supply a design static pressure of about 60 inches of water, which is approximately the required value for most commercial applications. Electric motors -- preferably three-phase induction motors -- normally would provide the power to drive the supercharging fans, although engines or mechanical connection to the turbine itself are possible alternatives.

While Figures 2A-2C show two supercharging fans, three or more supercharging fans can be used, or even a single supercharging fan may be used. Multiple supercharging fans allow for staging of fans to adjust turbine inlet pressure in increments, whereas a single supercharging fan does not provide this control option.

Preferably, the two supercharging fans have approximately equal pressure capability but are of unequal flow sizing. The lead supercharging fan has a larger flow capacity and preferably operates at a fixed speed to reduce cost. The lag supercharging fan has variable flow capacity and adjusts turbine inlet pressure as is described more fully below. Variable-speed drives or, in the case of axial fans, variable-pitch blades are preferred means for adjusting supercharging fan flow. Inlet vanes are another alternative,

but not preferred because of their relatively poor efficiency.

Figure 3 shows an alternate embodiment that uses an indirect evaporative cooler that can approach the ambient dewpoint temperature. The configuration is similar to that of Figure 2 except that an indirect evaporative cooler 60 is located in the air stream between the plenum 38 and the gas turbine 10. As with the previous embodiment, the gas turbine system 10 and generator 28 form a gas turbine power plant 11. Likewise, the gas turbine system 10 includes a compressor 12, a burner 14, and a turbine 16. The indirect evaporative cooler 60 uses a secondary air stream 62, which is taken from a portion of the primary air stream 64 that exits from an optional direct evaporative cooler 68 located between the indirect evaporative cooler 60 and the turbine 10 to optionally further cool the air entering the turbine. A turbine inlet air stream 66 is formed by the remaining portion of the primary air stream 64 and enters turbine 10. The air from the secondary air stream 62 is indirectly heated and humidified by the air flow from plenum 38 inside the indirect evaporative cooler 60 and exits as exhaust air stream 65.

Figure 4 shows another alternate embodiment that uses a motor-driven axial-flow supercharging fan. Supercharging fan 116 includes a motor 100 that drives impeller 102, both of which are contained in housing 106. The motor 100 is preferably a three-phase induction motor and is connected to a utility power line by conductors 110, 112, and 114 through switching contactor 104. When the contactor 104 closes, the motor 100 is energized, thereby driving impeller 102 which increases the pressure of discharge air stream 108 entering the turbine system 10. When the contactor 104 is open, the motor 100 is de-energized and the impeller 102 is driven solely by the air stream 108, thus

acting as a pressure reducer which reduces the pressure of air stream 108.

The contactor 104 may be a simple, manually operated device, in which case an operator will determine when extra turbine capacity is appropriate and close the switch to operate the supercharging fan. The preferred arrangement includes a thermostat 109 -- preferably located in contact with the air stream 108 -- which controls the contactor 104 and allows the supercharging fan 116 to operate when the temperature of the air stream 108 exceeds a predetermined value. Thermostat 109 thus functions to limit turbine output power and therefore prevents overloading of the gas turbine power plant 11.

More sophisticated controls are possible. For example, the supercharging fan can have variable-pitch blades, which are adjusted by a controller that senses pressure and temperature conditions and varies the pitch of the blades to maximize turbine output. Very sophisticated control is currently possible with microprocessor-based systems. Additional mechanical hardware can be added to improve performance. For example, a bypass damper can reduce the pressure drop through the supercharging fan when it is not energized. A direct evaporative cooler or other cooling means could be added to reduce turbine inlet temperatures.

One advantage of the system shown in Figure 4 is its low cost and simplicity, which are especially important in small turbines. The optimum configuration requires careful evaluation of the commercial application. While not preferred, the switch 104 may be eliminated in special applications. For example, if a turbine is moved from a cold climate to a tropical climate, it may be desirable to add a supercharging fan that runs continuously whenever the turbine is running. (Essentially, the supercharging fan would

adjust the design conditions of the turbine to match the higher ambient temperatures.)

Another, less preferred option would be to drive the supercharging fan directly from the turbine. The simplest configuration would be a direct mechanical connection using a shaft that is attached to the turbine. However, normally this arrangement would require a reduction gear to allow the supercharging fan to run at a speed that is much slower than that of the turbine. An eddy current clutch or mechanical clutch could be used to allow changes in fan speed. Another option would be to use a differential gearing to a brake, a generator, or other reactor to reduce the speed of the fan by controlling the speed of a second shaft. These arrangements would be difficult (or perhaps impossible) to retrofit onto existing turbines, and the gearing and other mechanical components necessary for this approach would require regular maintenance. Accordingly, this alternate embodiment may not be as reliable or desirable as others.

Figure 5 shows another alternate embodiment, this one having two supercharging fans arranged in series. A first supercharging fan 216 includes a first impeller 202 and a first motor 200 in a first housing 206. The first fan is located in an inlet air stream 208 which enters compressor 12. Conductors 210, 212, and 214 connect the first motor 200 through switch 204 to the utility power line. A second fan 236 is located upstream of the first fan 216. The second fan comprises a second impeller 222 and a second motor 220 in a second housing 226. The second motor 220 is connected through switch 224 and conductors 230, 232, and 234 to a utility power line.

Operation and benefits of the system of the invention will now be described in the context of Figures 2A, 2B, and 2C, which show a system that operates with two

supercharging fans arranged in parallel. In Figure 2A, both the first and second supercharging fans 30 and 32 are operating, and the bypass damper 36 is closed to prevent backward flow away from the turbine. First and second damper 42 and 44 are both open to allow air flow from the fans.

Figure 2B shows operation with the first supercharging fan 30 off and the second supercharging fan 32 on. The bypass damper 36 is again closed. The second damper 44 remains open, while the first damper 42 is closed. In Figure 2C, both supercharging fans 30 and 32 are off and their corresponding dampers 42 and 44 are closed. The bypass damper 36 is open to allow air to go around the supercharging fans 30 and 32.

Figure 6 illustrates the benefits of such a system. This figure is based on published performance data for gas turbines. The base system without any supercharging or inlet cooling has a peak power output around an ambient temperature of 40°F, and performance drops rapidly at higher temperatures. The figure shows that the capacity of the present invention is essentially flat, while the capacity for conventional systems drops rapidly at high ambient air temperatures. The base system is a simple-cycle turbine that gives a maximum output power of 100 MW below 40 °F, but much lower capacity at higher ambient temperatures. The present invention also has the same 100 MW capacity but maintains it at high ambient temperatures.

The bigger base and bigger base with evaporative cooler are simple-cycle turbines designed to match the present invention at high ambient temperatures. The bigger base also can be a supercharged turbine from the prior art. The bigger base with evaporative cooler takes advantage of the lower inlet temperatures available with evaporative cooling

to reduce the size requirements of the turbine for a given capacity at high ambient temperatures. The performance of the bigger base with evaporative cooling is also similar to that of supercharged turbines from the prior art.

Conventional evaporative inlet cooling helps the performance of the turbine at higher ambient temperatures without increasing the maximum output power at 40°F. A conventional supercharger, which includes an evaporative cooler, further increases turbine output power at all ambient temperatures, which would undesirably overload the gas turbine power plant at lower ambient temperatures.

The present invention limits turbine output to allow the benefits of supercharging at high ambient temperatures while preventing overloading of the power plant at low ambient temperatures. This new feature results in a power output that varies very little with ambient temperature changes.

Table 1 shows a cost comparison (adapted from Kolp et al.) for the supercharger of the present invention compared to conventional systems. This table shows that adding the new supercharger is less than half of the cost of adding peaking turbine capacity. The incremental cost per kW for adding the supercharger to a system with an evaporative cooler is about \$300 per kW, whereas the cost of a new peaking turbine power plant is \$700 per kW. The combination of an evaporative cooler and the supercharger increases turbine capacity at summer-peaking conditions by over 30%. The controls in the present invention eliminate the need for the bigger generator and related hardware, since the peak power output of the turbine at low ambient temperatures is unchanged. This analysis shows that the present invention has a significant advantage in new installations. In

retrofit situations, the cost of adding a conventional supercharger would be at least an order of magnitude larger since it would require replacement of the generator and related equipment in order to handle the increased output at low-ambient-temperature conditions.

Table 1: Cost comparison

	Incremental Cost	kW at Peak	Incremental Peak kW	Incremental Cost per kW
Peaking Turbine:				
Base	$21,260,400	30,372	30,372	$700
plus evaporative cooler	$220,000	36,180	5,808	$38
plus supercharger	$1,386,000	40,749	4,569	$303
plus bigger generator, etc.	$580,550	40,749	0	infinite
Combined Cycle Plant:				
Base	$38,885,000	38,885	38,885	$1,000
plus evaporative cooler	$220,000	47,415	8,530	$26
plus supercharger	$1,386,000	53,183	5,768	$240
plus bigger generator, etc.	$1,796,238	53,183	0	infinite

Table 2 shows how adding a supercharging fan can improve power plant performance. For the example in the table, a supercharging fan would increase the

turbine output by just over 4 MW while consuming only 1.24 MW of power. The result is an increase in net power output of almost 2.8 MW. This simple analysis shows that a supercharging fan can significantly improve peak power output for gas turbines.

Table 2: Power Comparison for a 100-MW plant

	Base	Cooler Only	Aux. Fan Only	Aux. Fan and Cooler
Inlet static pressure change (inches of water)	0	-0.5	10	9.5
Inlet temp. °F	95	77	99.9	78.7
Temperature change effect on turbine output (MW)	0	7.20	-1.96	6.5
Static pressure effect on turbine output	0	-0.30	6.00	5.70
Aux. Fan Power (MW)	0	0.00	-1.24	-1.24
Net change in turbine power output (MW)	0	6.90	2.80	10.96

(Assumptions: 8 CFM/kW, 95°F DB, 75°F WB, 80% fan efficiency, 95% efficient fan motor, 0.4% change in kW/(F, 0.6% change in kW/inch H_2O, 90% cooler effectiveness.)

This table also shows that, in combination, a supercharging fan and an evaporative cooler work synergistically to increase turbine power output. As stated before, adding the

supercharging fan to the turbine without a cooling mechanism would increase power output by 2.80 MW. On the other hand, adding the same fan to a turbine with a direct evaporative cooler adds 4.06 MW of capacity. The effect of the supercharging fan is thus almost 50% greater with the cooler than without the cooler. This analysis shows that the effect of the combination is greater than the sum of the parts and thus is especially desirable.

Table 2 shows that a significant improvement is possible with a supercharging fan with only 10 inches of static pressure. Additional improvement is possible for supercharging fans with higher static pressures. In many installations the optimum supercharging pressure may exceed 60 inches of static pressure.

Figures 7A and 7B are compressor maps illustrating the improvement in turbine capacity which can be obtained with the present invention. The vertical axis is turbine pressure ratio, which is the turbine inlet pressure divided by the atmospheric pressure. The horizontal axis is the mass flow parameter, which is given by the equation:

mass flow parameter = $\frac{m\sqrt{\theta}}{\delta}$,

where:

m is the turbine mass flow rate,

δ is the compressor inlet pressure divided by the standard atmospheric pressure, and

θ is the compressor inlet absolute temperature divided by design absolute temperature.

The pressure ratio is the compressor discharge pressure divided by the

atmospheric pressure. For purposes of this analysis, the effect of burner pressure drop and other minor factors are lumped with the compressor and turbine performance. (For background information in component matching, see Blathe, Fundamentals of Gas Turbines, chapter 10.)

Compressor curve 300 shows the performance of the compressor at design conditions, while compressor curve 301 shows the performance at peak inlet temperature. Turbine line 302 shows the performance of the turbine at design conditions, and turbine line 303 represents the turbine performance at peak inlet temperature. The intersection of compressor curve 300 and turbine line 302 defines the design operating point 304. The intersection of the compressor curve 301 and turbine line 303 is operating point 305 at the peak inlet temperature. The operating line 306 shows possible turbine operating points at different inlet temperatures. Surge line 307 is the limit of stable operation for the compressor.

At operating point 305, the turbine capacity is significantly reduced from that at the design operating point 304. The higher inlet temperature increases the speed of sound of the air, which reduces the compressor Mach number and moves the compressor curve to the left as shown in the figure. In addition, the higher temperature reduces air density which further reduces mass flow rate. These changes reduce the compressor pressure ratio and mass flow rate, which reduces the energy available for driving the turbine. Cooling the air can restore the turbine capacity.

Figure 7B illustrates how the new system can improve turbine capacity at peak ambient temperatures. Pressurizing the air entering the compressor increases the turbine

pressure ratio and compressor inlet temperature to create a compressor curve 310. A new turbine line 312 reflects the slightly higher temperature. The intersection of the turbine line 312 and the compressor curve 310 defines an operating point 314. This operating point corresponds to the operation with a supercharging fan and no inlet cooling. Operating line 318 shows possible operating conditions with different operating pressures. An additional benefit of higher pressure air is increased air density, which further enhances the capacity improvement.

Another compressor curve 316 and a turbine line 320 correspond to a lower compressor inlet temperature that can be achieved using an evaporative cooler. An operating point 322 at the intersection of the compressor curve 316 and turbine line 320 corresponds to an operating condition for a supercharging fan and inlet cooling. This analysis indicates that it is possible to approximate the original design capacity of a gas turbine through a combination of inlet cooling and pressurization. The ultimate limits of the turbine capacity are the operating pressures and power output that are acceptable for the turbine, generator, etc. These factors would normally prevent the supercharging fan and inlet cooling from creating a turbine output that is significantly above the turbine design output.

Figure 8A plots fan curves showing how parallel supercharging fans can work together to create a range of turbine inlet pressures. For parallel operation, the pressure across the two fans is the same and the flows add together. A lead fan curve 350 is for the lead fan. The gas turbine line 356 is nearly vertical since the flow through the turbine varies only slightly with inlet pressure. The operating point 358 is at the intersection of

the lead fan curve 350 and the turbine line 356. This operating point 358 corresponds to operation with one fan.

A first lag fan curve 352 corresponds to the performance of the lag fan at full speed. Fan curve 364 corresponds to running both fans together. The intersection of the fan curve 364 and the turbine line is an operating point 366 that corresponds to operation with both fans. A second lag fan curve 354 represents fan performance at low speed, and fan curve 360 represents the corresponding two fan operation. An operating point 362 represents the operating condition with both fans operating and the lag fan at low speed.

Figure 8B illustrates operation with two similar fans in series. Fan curve 372 corresponds to operation with only one fan. The intersection of the fan curve and the turbine line 370 represents operation point 378 for one fan. Fan curve 374 corresponds to both fans running. Operating point 376 represents turbine operation with both fans running.

Figure 9 shows a simple relation between maximum supercharging pressure and inlet temperature that may be used to control the supercharger. The controller can use the temperature entering the compressor to adjust the supercharger pressure. The result is a very simple control system for maximizing turbine performance. This approach may be very useful in controlling a supercharger in retrofit situations since it would require little or no changes to existing controls for the gas turbine power plant.

More sophisticated control and operation are also within the scope of the invention. For example, another preferred embodiment having a recirculation arrangement around the compressor to control capacity of a gas turbine power plant 421

is shown in Figure 10. The gas turbine power plant comprises a gas turbine system 432 and a generator 426. The gas turbine system 432 includes burner 422, compressor 420, and turbine 424, the latter two of which share a common shaft 430. Compressor inlet air stream 440 enters the compressor 420 and is compressed to form burner inlet air stream 442. The burner 422 heats the air stream 442 to form burner outlet air stream/ turbine inlet air stream 444 that enters turbine 424. The turbine 424 extracts power from the air stream, which exits as exhaust 446. The gas turbine power plant also includes structure, bearings, controls, and other components which are known in the art and therefore not shown. The gas turbine power plant also may include a bottoming steam cycle system or multiple shaft arrangements.

The turbine 424 drives the compressor 420 and generator 426, which also shares the shaft 430. The generator supplies electric power to the utility grid through conductors 436 and a transformer 428.

Supercharging fan 423 pressurizes fan inlet air stream 453 to form a pressurized air stream 455 that enters a first evaporative cooler 425. The first evaporative cooler 425 includes a pump 437 and an evaporative pad 435. The first evaporative cooler 425 cools the pressurized air stream 455 to form compressor inlet air stream 440.

The supercharging fan supplies a static pressure on the order of 60 inches of water and preferably may be a centrifugal fan or an axial fan. The pressurized air stream 455 and the compressor inlet air stream 440 are confined in ducts that can handle this high pressure; preferred duct configurations are described below in connection with Figures 15, 16A, 16B, 17A, and 17B.

A second evaporative cooler 470 is provided upstream of the supercharging fan 423. The evaporative cooler includes an evaporative pad 434 and a pump 436, which circulates water over the pad to create a wet surface for cooling the ambient air stream 451 by means of evaporation before it is supplied to supercharging fan 423 as fan inlet air stream 453. The evaporative coolers each also may include a sump with a float valve to control water level and a means for bleeding off a small portion of the circulated water to prevent build-up of salts.

Numerous evaporative coolers are available commercially from a variety of manufacturers, so details of the cooler design are not included. Additionally, while direct evaporative coolers are illustrated, indirect evaporative coolers or indirect-direct evaporative coolers also may be employed.

A key feature of the present invention is the relative sizing of the supercharged gas turbine and the generator and associated equipment. In particular, the generator and the supercharged turbine are sized so that the generator operates at nearly full capacity at summer-peaking conditions. (In contrast, according to the prior art designs, the generator and auxiliaries would be sized based on full supercharged output at winter conditions, which would typically be inlet temperatures of 40°F or lower, and performance would "fall of" from these as ambient temperatures increased.) Accordingly, the present invention includes means for controlling turbine capacity at low ambient temperatures so as to prevent overloading the generator.

Preferably, a controller 460 receives a current signal 468 from a current sensor 462 that senses generator current. The controller 460 preferably includes the normal

capacity and safety functions of the gas turbine power plant, but it may alternately be a stand-alone unit. The current sensor is preferably a current transformer, in which case the signal is in the form of an AC current. Other possible sensors may be such as to provide a voltage, optical, or radio frequency output signal or some other type of signal.

The controller 460 provides a damper control signal 464 to damper 450 to control the flow of a heated air stream 452, which is drawn from the burner inlet air stream 442 and circulates through the damper 450 to the compressor inlet. The controller 460 also provides pump control signals 466 and 467 to pumps 436 and 437, respectively, and a fan control signal 465 to the supercharging fan 423. Pump and fan control for this embodiment may be as simple as on/off control, although variable control with variable-speed drives, inlet vanes, or variable-pitched blades also might be employed.

Because the system is sized to provide maximum turbine/generator performance at summer-peaking temperatures, as ambient temperature drops and the turbine output increases, generator output will start to rise and exceed the maximum design output. Therefore, as the ambient air temperature drops, the current sensor 462 senses a correspondingly higher generator current signal 468 and communicates this to the controller 460. The controller responds first by turning off the pump 436, thereby deactivating the second evaporative cooler 470 and allowing the fan inlet air stream 453 to approach the ambient dry-bulb temperature. If the ambient temperature drops further, the preferred control response is to start to open the damper 450 to allow heated air 452 to flow from the compressor outlet to mix with the compressor inlet air stream 440. As ambient air temperature drops even further, the damper will open even further so as to

limit the generator current to that produced at summer-peaking performance.

If the ambient temperature drops even further, the controller 460 sends a fan control signal 465 and a pump control signal 467 to turn off the supercharging fan 423 and the pump 437 for the first evaporative cooler 425; these signals terminate supercharging entirely. The controller 460 may then send a damper control signal 464 to close the damper 450.

Another preferred alternate embodiment of a turbine system according to the invention is shown in Figure 11. Similar to the embodiment shown in Figure 10, a controller 474 receives a current signal 468 from a current sensor 462 that senses generator current, and the controller 474 provides a pump control signal 466 to pump 436 in response thereto. Rather than regulating a damper to provide heated compressor output air back into the compressor inlet, however, the controller 474 also provides a burner control signal 472 to the burner 422, which regulates the burner output and hence turbine output. Preferably, the normal operating and safety controls for the gas turbine power plant are integrated into the controller 474, but the controller also can be a stand-alone controller.

The control approach used with the embodiment shown in Figure 11 is similar to that used with the embodiment shown in Figure 10. As the ambient temperature drops and causes a resultant increase in output generator current, controller 474 responds by turning off pump 436 to deactivate the second evaporative cooler 470. If the generator current still exceeds the summer-peaking design output, the controller 474 responds by adjusting the burner control signal 472 to reduce the output of burner 422. At still lower

ambient temperatures, the controller will turn off the supercharging fan 423 and the first evaporative cooler 425 altogether so that the power plant operates without any supercharging at all.

Figure 12 shows another alternate embodiment of the invention in which turbine output is controlled by regulating compressor inlet temperature using a heater. Controller 484 receives a temperature signal 490 from a temperature sensor 488 that is located in the compressor inlet air stream 440. The controller 484 provides a heater control signal 482 to a heater 480 located upstream of the compressor inlet, and the heater provides a heated air stream 486 to the compressor 420. (While the preferred location for the heater is upstream of the evaporative cooler, the heater alternatively may be located between the evaporative cooler 470 and the compressor 420 if the heater is made of materials that can handle high relative humidity without excessive corrosion.)

There are a number of options for implementing the heater. One simple option is to use a gas burner. A second option is to use a boiler with a separate liquid-to-air heat exchanger. A third option is to use a heat exchanger that recovers heat from the turbine exhaust 446. (This option should provide the best efficiency and therefore is preferred if installed cost is not excessive.) Electric heaters are a fourth option, although they are not preferred because of their poor efficiency. Finally, blowing a portion of the exhaust 446 into the compressor inlet air stream 440 is a low-cost fifth option, but that may cause corrosion problems in the compressor 420 or other components. Regardless of the specific type, the heater should be capable of modulating its output so as to maintain the heated air stream 486 at an approximately constant temperature.

The control approach utilized with the embodiment shown in Figure 12 is to maintain a minimum temperature of the supercharging fan inlet air stream 453. Therefore, as the ambient air temperature drops, the controller 484 responds by turning off the pump 436 to deactivate the evaporative cooler 470. If the ambient temperature drops further, the controller 484 provides a heater control signal 482 to turn on heater 480 and adjust its output to maintain the required temperature of the fan inlet air stream 453.

Figure 13 shows another, similar alternate embodiment that is especially suitable for use with a combined-cycle power plant. A combined-cycle gas turbine power plant 506 includes an additional steam cycle system 498 that utilizes exhaust heat from the turbine 424 to generate additional power. The steam cycle 498 includes a boiler 504, a steam turbine 500, a condenser 502, and a feed water pump 503, which are all connected together in a circuit. A liquid-to-air heat exchanger 491 is provided in a fluid loop with pump 492 and heat recovery heat exchanger 496. The pump 492 circulates heat transfer liquid 512 through the heat recovery heat exchanger 496, where it absorbs heat from exhaust stream 508. The pump receives a signal 494 from controller 484, which regulates flow of the heat transfer liquid 512, thereby controlling the temperature of heated air stream 486 entering the evaporative cooler 470. As with the embodiment in Figure 12, the controller also can turn off the second evaporative cooler 470 as a first step in controlling turbine inlet temperature, and then ultimately may shut down supercharging altogether.

(One concern with using the exhaust stream 508 as a heat source, however, is corrosion, and corrosion-resistant materials are required to handle the presence of nitric

acid and possibly sulfuric acid. One possibility is to use plastic materials for the various conduits, but plastics usually have relatively low temperature limits. Therefore, it may be desirable to mix the exhaust with ambient air to reduce the maximum temperatures, thereby permitting use of plastics. Direct contact liquid-to-gas heat exchange is also an option. The liquid would contain a suitable neutralizing agent (such as sodium bicarbonate) to prevent problems with acidic condensate. Here again it may be desirable to mix the exhaust air stream with ambient air to reduce maximum operating temperatures.)

In addition to an increased overall generating capacity at high ambient temperatures, the system configuration shown in Figure 13 also should provide enhanced efficiency, since combined cycle power plants actually increase their efficiency slightly at higher compressor inlet temperatures. Moreover, whereas Figure 13 illustrates a heat transfer loop for recovering waste heat, many other configurations are possible. For example, heat from the condenser 502 could be used to warm the inlet air stream. Providing an air-to-air heat exchanger between the exhaust and inlet air streams is also an option.

The illustrated embodiments described above represent possible configurations of the present invention. While these embodiments use either generator current or compressor inlet temperature as feedback control parameters, other parameters may be used to achieve similar results. For example, control inputs may include generator power; ambient dry-bulb temperature; ambient wet-bulb temperature; shaft torque; or other inputs.

Furthermore, while Figures 10 through 13 show a second evaporative cooler located upstream of the supercharging fan, this feature is optional and can be eliminated without a major change in performance. The second evaporative cooler does enhance the pressure capability of the supercharging fan at high ambient temperatures, however, and thus adds some additional output power.

The use of the first evaporative cooler is also optional to some extent, but performance will suffer significantly if it is eliminated. Therefore, only in cases where water and/or space is/are severely limited would it be desirable to eliminate the first evaporative cooler.

Finally, with respect to the various embodiments described above, simple on/off control is the preferred method of control for the supercharging fan. More sophisticated control using variable-speed drives or variable-pitch fan blades is an option, however, and may result in some energy savings, but controlling the burner or inlet temperatures already provides a means for modulating power output.

Figure 14 illustrates a preferred embodiment that incorporates an indirect evaporative cooling system and a variable-speed drive for the supercharging fan. As illustrated in Figure 14, a variable-speed drive 550 receives electrical power from the utility grid and supplies variable-frequency AC power to an induction motor 522. The induction motor 552 drives a shaft 554 that drives a supercharging fan 556. The supercharging fan draws ambient air and supplies a pressurized air stream 558 to a cooling coil 560. The cooling coil is a water-to-air heat exchanger that cools the air without adding moisture to it. A cooled air stream 570 exits the coil and enters an

evaporative cooler 425, which is configured as described above.

The cooling coil 560 is connected, by means of piping 562, to a pump 564 and a cooling tower 566 to form a circuit, which circuit acts as an indirect evaporative cooler. The cooling tower is preferably a forced-draft wet tower which can cool water to temperatures approaching the ambient wet-bulb temperature.

(In areas where suitable water supplies are limited, an alternative would be to use a dry tower and eliminate the evaporative cooler. In that case, the dry cooling tower and coil assembly would act as a simple heat exchanger between the ambient air and the air leaving the supercharging fan.)

Other heat exchanger and cooling configurations are contemplated. For example, the cooling tower and cooling coil could be replaced with an air-to-air heat located exchanger between an ambient air stream and the air exiting the supercharging fan. In that case, a direct evaporative cooler could be placed upstream of the heat exchanger, on the ambient air side, to provide additional cooling. While these configurations are possible, they are not preferred because the large required heat exchanger area and the large required pressure differential between the two air streams would create the need for a large, expensive heat exchanger.

In climates where freezing presents a potential problem, the piping, cooling coil, and cooling tower would require suitable freeze protection. Possible protection alternatives include insulation and heaters, drain-down provisions, and the use of brine in the cooling coil with a water-to-brine heat exchanger. (A water-to-brine heat exchanger would provide the additional benefit of isolating the cooling coil from the fouling dirt

associated wit the cooling tower; an appropriate filter could be included to reduce fouling in systems without a secondary loop.)

In this embodiments, controller 568 prevents overload of the gas turbine at lower ambient temperatures. The controller 568 receives a temperature signal 572 from a temperature sensor 573 located at the compressor inlet, in the compressor inlet air stream 440. The controller also receives a pressure signal 574 from a pressure sensor 575 located in the pressurized air stream 558. The pressure sensor 575 can be located anywhere in the air stream between the supercharging fan 556 and the gas turbine system 432, since the pressure changes are small in this section.

The controller 568 provides a speed control signal 578 to the variable-speed drive 550. This signal modulates the speed of the supercharging fan to maintain the optimum supercharging pressure. The controller also provides an output signal 576 to pump 564 in the cooling tower loop and an output signal 580 to the pump 437 in the evaporative cooler. These two output signals would normally provide a simple on/off control.

While this embodiment uses a variable-frequency drive to control fan speed, there are many other possible ways of modulating fan output that could be used. Examples include electromechanical variable-speed drives such as described in U.S. Patent No. 5,947,854 and co-pending provisional application serial number 60/164590; eddy current clutches; DC motors; variable pitch fan blade; variable inlet vanes; etc.

An important advantage of this embodiment is its ability to supply relatively cooler air to the gas turbine system. The wet bulb temperature of the air entering the cooling tower is unaffected by the energy input from the supercharging fan. Therefore,

the cooling coil can cool the airstream 558 to temperatures approaching the ambient wet-bulb temperature without adding any moisture. The evaporative cooler 425 can then further cool and humidify the air stream. As a result, the airstream entering the turbine system can be 10°F cooler than in the case where a simple direct evaporative cooler is employed. This reduction in temperature provides roughly an additional 2.5% increase in power output from the gas turbine. Because the cooling tower fan and pump typically consume only 10% of this capacity increase, a net capacity improvement of over 2% can be obtained with this system.

More complicated cooling systems are also possible. For example a mechanical, desiccant, or absorption cooling systems may replace the direct evaporative cooler. Cooling systems using ground water or cold lake or ocean water are another option. These systems can achieve much lower air temperatures, which can further enhance the capacity improvement. The disadvantage of this approach is the additional complexity and cost of the additional cooling system.

As noted above, the pressurized air streams and the compressor inlet air stream are confined within ducts able to withstand relatively high pressure, and Figure 15 shows details of the preferred embodiment of such ductwork. An interior duct 610 is disposed inside a round, outer duct 611, and a flow passage 612 equalizes the pressure of a fluid flowing inside the interior duct and a space 615 that is located between the two ducts. (For the present application, the fluid would be air, but the ductwork assembly may be used to convey any high pressure fluid.) Preferably, the space 615 is filled with fluid or may include porous material such as fiberglass, open-cell foam, etc. One advantage of a

porous fill material is that it can help to reduce noise and also helps to support the ducts.

The flow passage 612 for equalizing pressure across the interior duct may be simply an aperture or it can be a series of apertures or constituted by a porous surface. The flow passage may also be provided by cracks or other small openings that are common in unsealed ductwork. The critical requirement for the design of the flow passage 612 is that it must be able to permit sufficient fluid flow rates through it to ensure that if a leak develops between the round outer duct and the ambient atmosphere, an excessive pressure drop across the interior duct will not result. Advantageously, the flow path may include a pressure relief valve to reduce the risk of damage to the interior duct if such a leak does occur.

While Figure 15 shows a rectangular interior duct, the interior duct may have virtually any shape other than a round shape, and it may include large, flat surfaces without concern for excessive strength requirements. The interior duct need only handle any pressure drops, velocity pressure, turbulence, etc. associated with fluid movement through the duct.

The outer duct preferably has a circular cross-section to minimize material requirements. Other shapes such as oval or elliptical may be used, but they are not preferred since they would increase the strength requirements significantly. Large round ducts may include corrugations or other reinforcements to improve rigidity and reduce risk of damage to the duct from wind or accidental loading.

Typical materials for constructing the ducts include metals such as steel or aluminum. Other possible materials include plastic, wood, ceramics, etc. Material

selection depends on factors such as strength, cost, and compatibility with the particular fluid flowing through the ducts.

Figure 16A and 16B show a front and a cross-sectional view, respectively, of a transition duct which may be used to connect two different sized ducts. A circular, conical outer duct 620 encloses a tetrahedral or pyramidoidal interior duct 621. A flow passage 622 is provided in the wall of the interior duct 621 to equalize fluid pressure between the inside of the interior duct 621 and the space 623 that is located between the two ducts. This transitional duct assembly can be used either as a diffuser or a flow accelerator, depending on the direction of fluid flow. Moreover, although Figures 15, 16A, and 16B illustrate basic configurations of the high pressure duct according to the invention, this invention can be applied to practically any particular duct geometry, including elbows, tees, transitions, etc.

Figures 17A and 17B illustrate such ductwork employed specifically with a gas turbine supercharger. Figure 17A shows the cooler, without a supercharger, with a direct evaporative cooler 630 at the end of a rectangular duct 631 that supplies air to the inlet of a gas turbine. The rectangular duct 631 is designed to carry a pressure difference (as compared to ambient) of just a few inches of water or less.

Figure 17B shows the corresponding supercharged configuration. The fan 635 increases the air static pressure to very high levels -- typically about 60 inches of water. The fan 635 is connected to an end piece 639 of a diffuser duct 636. The diffuser duct is connected to a straight duct 637 that encloses the evaporative cooler and a contraction duct 638. Each of these ducts includes a round duct on the exterior that encloses a

rectangular interior duct.

This ductwork arrangement provides several significant advantages over the prior art:

- Lower material weight and cost compared to conventional rectangular ducts;
- Lower pressure drop than conventional round ducts used with rectangular components;
- Simple geometry for easy construction; and
- Ability to retrofit lower-pressure ductwork to withstand higher pressures.

These advantages are especially desirable for use in gas turbine superchargers. The high pressures (approximately 60 inches of water) and the large duct dimensions (duct diameter of 30 feet or more) create the need for an extremely strong rectangular duct that requires huge amounts of material for reinforcement. The new duct configuration of the invention eliminates the need for the rectangular duct to withstand the high gas pressure, which greatly reduces the material cost and weight. It also allows the use of existing evaporative coolers and associated ducts, which greatly reduces installation cost of the supercharger. While the invention is especially attractive for use in gas turbine superchargers, it can be used in air-conditioning systems and other industrial and commercial applications that require moving gases or liquids.

Fig. 18 illustrates an alternate preferred embodiment of the invention, using a fogger in the inlet air stream of the gas turbine. A gas-turbine power plant 121 comprises a compressor 120 and an expander 124 that are rigidly attached to a shaft 130 that drives a generator 126. An air stream 191 enters the compressor, which pressurizes the air and

supplies it to a combustor 122. The combustor heats the air and supplies it to the expander 124. The expander extracts work from the expanding gas to drive the compressor and the generator.

A supercharger 190 is located upstream of the gas-turbine power plant. The supercharger comprises a fan 140, a first fogger 149, and a second fogger 169 that are located inside a duct 147. The first fogger is located upstream of the fan, while the second fogger is located between the fan and the turbine.

The fan 140 comprises a hub 141 and fan blades 142. The fan is rigidly attached to a motor shaft 144. A motor 146 drives the motor shaft 144 and thereby drives the fan 140. The fan is preferably a variable-pitch axial flow fan. The hub 141 includes a mechanism for adjusting the pitch of the fan blades 142 to adjust fan output pressure and flow.

The motor is preferably a three-phase induction motor or other electric motor. Another option is to drive the fan directly from the main turbine, which eliminates the need for the motor. Separate prime mover for the fan, such as a second gas, a steam turbine, or an internal-combustion engine, is also an option, though not preferred. An important advantage of an electric motor is that it is relatively easy to install on an existing gas turbine.

The output of the fan is on the order of 60 inches of water static pressure. The optimum pressure depends on the availability of a suitable fan, generator capacity, turbine capacity, and other factors.

A multistage, axial-flow fan, as shown in Figure 18, can achieve this static

pressure. Centrifugal fans or single-stage axial fans are also an option. If centrifugal fan is used, variable-pitch blades are not normally an option so a variable-speed drive is the preferred means for controlling fan capacity. Other options include variable inlet vanes or dampers, but they are less efficient. Variable-speed is also an alternative for axial fans.

The first fogger 149 comprises a first manifold 156, second manifold 158, and a third manifold 160. Each manifold has spray nozzles that create mist 162. The first manifold receives pressurized water from a first pump 150. Likewise a second pump 152 and a third pump 154 supply pressurized water to the second and third manifolds 158 and 160 respectively. The pump outlet pressure is preferably roughly 1000 to 3000 psi. A stream of water 164 feeds the pump inlets. The water is preferably filtered, demineralized water. An air stream 148 is drawn into the duct 147 through the first fogger 149.

The second fogger 169 is located downstream of the fan. Like the first fogger, the second fogger is comprised of multiple manifolds and pumps. The fourth, fifth and sixth manifolds, 176, 178, and 180 are connected to fourth, fifth, and sixth pumps 170, 172, and 174 respectively. The action of pressurized water in the nozzles in the manifolds creates a mist 182.

There is a large degree of flexibility in the design of the foggers. For example the number of manifolds in each fogger is somewhat arbitrary. A larger number allows for easier control over the amount of fog produced and provides additional redundancy. On the other hand, fewer manifolds simplify installation and may reduce cost. In addition manifolds are not necessarily of equal capacity.

As far as the capacity of the foggers, the first fogger is preferably sized to ensure nearly saturated air on the outlet of the fan. The second fogger would further saturate the air and provide extra moisture for cooling inside the compressor 120. The total mass of water added to the air stream would preferably be the amount for saturation at the inlet to the compressor plus roughly 0 to 2% of the air mass flow rate.

Sub a' A controller 161 controls the operation of the supercharger 190. The basic approach is to reduce fan pressure and the amount of fogging at lower ambient temperatures to prevent overload of the generator and other components in the gas-turbine power plant. A fan inlet temperature sensor 182 and a fan outlet temperature sensor 184 provide input to the controller. As the ambient wet-bulb temperature drops, the fan inlet temperature provides a signal to the controller to reduce fan capacity by providing an output signal to reduce the pitch of the fan blades. In addition, the lower temperatures mean that less water is required to saturate the air, so the controller 160 can turn off some of the pumps for the foggers.

As temperatures approach freezing, the pumps for the first fogger can be turned off to prevent ice formation. The second fogger may still operate at this condition, if the capacity of the generator and other components is adequate.

At very low temperatures when no supercharging is possible, the fan and the foggers may be turned off and the fan may be allowed to free rotate in the air stream. A bypass damper around the fan may be provided to reduce pressure drop to the turbine under these conditions.

There are numerous variations on this basic embodiment. For example the second

fogger may be eliminated in cases where the turbine compressor is especially sensitive to the droplets of water. In this case the controller could modulate the amount of fog from the first fogger to ensure complete evaporation of the water droplet before they reach the turbine.

Another option, though not preferred would be to eliminate the first fogger. This change would cause only a relatively small penalty in performance if the capacity of the second fogger were increased to compensate.

Many options for the fan selection are possible. For example, multiple fans may be desirable for some applications. Multiple fans can provide redundancy to improve system reliability. They may also reduce cost through the use common parts and may allow for more sophisticated control options.

Fixed fan output is another simple control alternative. This approach is an option for cases where the generator is sized to handle the full output of the turbine at the normal operating conditions. This option is also possible in cases where another means of turbine capacity control, such as modulation of the combustor output or means for heating the inlet air stream, can prevent overload of the generator and other components.

As with conventional gas-turbine power plants, filters and silencers are normally provided with this system. The operation of the fan and the foggers is not normally affected by dust so the location of the filter is primarily a matter of convenience. Fog droplets do somewhat increase the pressure drop through filter, so the preferred location is normally upstream of the fogger. A silencer is preferably located upstream of the fan to prevent radiation of noise.

This system can supply a large capacity increase. For a conventional supercharger with a conventional evaporative pad with a 90% effectiveness and 60 inches of water supercharging, the capacity increase is between 20 and 30 percent. Fog intercooling can provide up to 10 percent additional capacity in additional capacity. In addition the fog can effectively provide 100% evaporative cooling effectiveness and reduces the theoretical fan power requirements by several percent. The result is a system that can readily achieve a capacity increase of 20 to 40% or more, depending on the climate and specific design limit of the gas-turbine power plant.

Advantages of This Embodiment

This preferred embodiment of the present invention has several important advantages:

1) A large increase in capacity: The system can achieve a capacity increase of 20-40 percent with most gas turbines.

2) Low cost: The cost of the fan and the fogger system is much less than that for new turbine capacity.

3) Compact: The elimination of the large evaporative pad required for conventional superchargers greatly reduces the size and cost of evaporative cooler and associated ductwork.

4) Easy retrofit: The small size and the ability to use the existing generator allow the system to be installed on existing turbines.

5) Improved controls: The controls allow the system to match the maximum

capacity of the gas-turbine power plant at a wide range of ambient temperature conditions.

6) Reduced fan power: The first fogger creates a mist that cools the air as it goes through the fan, which reduces the theoretical fan power required for a given pressure increase and mass flow rate.

7) Simplicity: The invention uses only a few simple components.

8) Reliability: The system uses proven components with good reliability. In addition the gas-turbine power plant can continue to operate without the supercharger in case of a component failure or other problem.

The advantages of the supercharging system for gas turbines, as a whole, are significant and numerous. Among the most important ones are:

- Large increase in turbine capacity at high ambient temperatures;
- Ability to achieve a large capacity increase even in humid climates;
- Low installed cost;
- Simple design;
- Compact design;
- Flexible control possible;
- Can be retrofitted on existing gas turbines;
- Bypass damper allows turbine operation without supercharging fan;
- Multiple fans and dampers provide redundancy for reliable operation; and
- Blower and cooling means work synergistically to give large capacity improvement.

Overall, this system represents a major breakthrough in gas turbine technology. Its simplicity and low-cost make it extremely desirable for power-generation applications that now face major performance penalties at high ambient temperatures. The invention having been thus described, it will become apparent to those skilled in the art that the same may be varied in many ways without departing from the spirit and scope of the invention. Any and all such modifications are intended to be covered within the scope of the following claims.